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A COMPARISON OF ANALYTICALLY AND EXPERIMENTALLY DETERMINED ISOTHERMAL PRESSURE LOSSES IN A HEAT-EXCHANGER INSTALLATION

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ADVANCE RESTRICTED REPORT

A COMPARISON OF ANALYTICALLY AND EXPERIMENTALLY DETERMINED ISOTHERMAL PRESSURE LOSSES IN A HEAT-EXCHANGER INSTALLATION

By Wesley H. Hillendahl

SUMMARY

This report contains analytically and experimentally determined values of the pressure losses in a heat-exchanger installation. The analytical values of pressure loss were determined by adding together losses calculated from published data on elbows, diffusers, and straight ducts. Good agreement was obtained between the analytical and experimental values of losses in the over-all and component parts of the installation, indicating that such an analysis is practical in predicting the pressure losses in ducting systems of this type.

INTRODUCTION

The principles of ducting design have been established on the basis that in turbulent flow the major sources of loss are skin friction and flow separation, and that these losses are roughly proportional to the dynamic pressure of the air in the duct.

A bibliography containing most of the reports in the field of ducting design is presented in reference 1. The majority of this material deals with pressure losses in isolated elbows and diffusers. These data are useful as a guide to the design of efficient ducting systems and provide a basis for the prediction of pressure losses in such installations.

As pointed out in reference 1, such application of the data has little experimental verification and the question arises as to the degree of accuracy with which a summation of

the calculated component pressure losses will yield the overall pressure loss in a system where the magnitude and location of interference effects are largely unknown. Interference effects are defined as those occurring where disturbances in one component cause the flow distribution at the entry of another component to be unsymmetrical, thereby affecting the pressure loss.

This report contains an analysis in which the material in the bibliography of reference 1 was used to calculate the pressure losses in a heat-exchanger installation. The isothermal pressure losses in the installation were measured for the purpose of providing an experimental verification of the analytical predictions. The comparison indicates the validity of the analysis and the magnitude of the interference effects in the test installation.

DESCRIPTION OF THE HEAT-EXCHANGER INSTALLATION

Heat exchangers of the type reported herein have found particular application in the field of ice prevention and cabin heating where they are used to supply an adequate and reliable source of heated air by utilizing the waste heat in the engine exhaust gas. The heat-exchanger installation herein described was developed for use in the ice-prevention system of a large airplane. The heat exchanger is located in the exhaust-gas ducting between the engine and two turbosuperchargers which are arranged for single or parallel operation. Turbosupercharger 1 is blocked off by a butterfly valve when turbosupercharger 2 is operating alone. When both are operating in parallel, the flow of exhaust gas is adjusted so as to be divided equally between the turbosuperchargers.

General views and construction details of the heat exchanger are shown in figures 1 and 2, respectively. In a cross-flow plate-type heat exchanger, such as the one tested, the exhaust-gas and air flow at right angles in alternate passages between formed steel plates, each plate thus forming an interface through which heat is transferred from the exhaust gas to the air. At design conditions of 155 miles per hour and 18,000 feet pressure altitude, a total of 800,000 Btu per hour is to be transferred from 12,000 pounds per hour of exhaust gas to an equal amount of air.

The headers shown in figures 3 and 4 serve as transition pieces in ducting air and gas into and out of the heat

exchanger. The branches in the exhaust-gas outlet header lead to the two turbosuperchargers.

ANALYSIS

The isothermal pressure losses in each component of the test installation were calculated as follows: The shape of the component was compared to those on which data (usually shown in graphs) were published, and where similarity existed the data were applied to the component. The loss factor corresponding to the dimensions of the component was then read from the reference data and multiplied by the local velocity head to obtain the pressure loss. Where losses in a component resulted from a combination of effects (i.e., diffusion, turning, contraction) each effect was isolated, the corresponding loss calculated, and the results added to give the total loss.

Air-Side Losses

Inlet-header losses.- The air inlet header is shown in figure 3(a) to consist of a vaned elbow superimposed on a diffuser. The loss in the elbow was computed using data on vaned elbows published in reference 2 where the pressure-loss factor is plotted as a function of the s/c or gap/chord ratio in the vanes. This value ranged from 0.48 to 0.62, and the corresponding loss is approximately 25 percent of the velocity head at the header entrance. At the design flow rate of 12,000 pounds per hour, the velocity head is 0.90 inch of water and the turning loss is 0.225 inch of water.

The loss caused by the expansion in area was computed using data on straight diffusers which appear in reference 3. If the expansion is assumed to occur principally in the vanes, an effective angle of divergence, based on the length of the vanes (3 to 7 in.) and the ratio between the inlet and outlet areas ($A_2/A_1 = 2.3$), has a value between 11° and 23°. The corresponding loss factor is about 20 percent of the change in velocity head through the vanes, which at the design flow rate is 0.73 inch of water. The expansion loss is then 0.145 inch of water.

The total air-inlet-header loss is then the sum of the turning and diffuser losses - or 0.37 inch of water.

Heat-exchanger losses.- The loss in the heat exchanger was assumed to result from friction on the plate surfaces and expansions in the three corrugations running transverse to the air flow and in the outlet end of the plates. (See fig. 2.) The friction loss was calculated using the Fanning equation (reference 4) and is 1.12 inches of water at the design flow rate.

Since the expansions have an included angle larger than 50°, the losses are approximately equal to those in sudden expansions. A formula for such losses appears in reference 3 and in this case the total loss for all four expansions is 2.02 inches of water.

The total heat-exchanger loss is 3.14 inches of water at the design flow rate.

Outlet-header losses.- Since the contraction is gradual, the principal loss occurring in the outlet header was assumed to result from the elbow. Data in references 2 and 5 were used. For a value of s/c of about 0.3, the loss factor is 0.2. As a check, if the vanes are omitted, the loss corresponding to a mean-elbow radius/diameter ratio R/D of 1 has a value of 0.25. This is considered to be good agreement and the loss corresponding to a velocity head of 0.85 inch of water is about 0.17 inch of water.

Over-all loss.- By adding together the above calculated values a result of 3.68 inches of water is obtained for the pressure loss in the air side of the installation at design flow rate.

The calculated values of pressure losses in the air-side ducting are plotted in figure 5.

Gas-Side Losses

Inlet-header losses.- As the air (during isothermal tests) enters the inlet header (fig. 4(a)), it is immediately turned through approximately 90° in an elbow having a circular cross section. Following the turn it passes through an expansion in duct area. An elbow which is followed by a straight section of duct causes a lower pressure loss than one which is not, because of recovery in the duct. This effect is shown in reference 5 for rectangular ducts. The same reference presents data only on circular elbows which are followed by straight ducts; therefore, in order to apply

these data to the gas inlet header, the loss was increased by the comparable difference between the curves for rectangular ducts.

Thus, for an R/D of 0.85, the elbow loss factor is 0.33, and by adding 50 percent (from comparison with rectangular ducts) the loss becomes 50 percent of the velocity head - or 0.52 inch of water.

Similar to the above calculation, the effect of the elbow on the diffuser loss is expected to be appreciable in this case. Because of the separation of flow at the inner radius of the elbow, the air is not diffused uniformly, and it is expected that the use of data on straight diffusers will result in an underestimate of the loss. In this case the loss was assumed to be equal to that in a sudden expansion. The loss was calculated to have a value of 40 percent of the change in velocity head, or about 0.35 inch of water.

The total calculated loss in the inlet header is then the sum of the two losses, or 0.87 inch of water at the design flow rate of 12,000 pounds of air per hour.

Heat-exchanger losses.- In effect, there are 88 parallel passages through the gas side of the heat exchanger, since the 22 passages between the plates are each divided into four by the corrugations shown in figure 2. It is assumed that the flow is equally divided between the passages, and that the calculation of the pressure drop through any one passage is the same as that of the entire exchanger.

Using the Fanning equation, as for the air side, the friction pressure drop was calculated to be 0.56 inch of water at the total flow rate of 12,000 pounds of air per hour through the heat exchanger.

The only expansion loss occurs at the outlet end of the passage and, as before, its loss is considered to be equal to that of a sudden expansion. Since the area ratio is 2.3, this loss amounts to 0.28 inch of water.

The sum of the heat-exchanger losses is then 0.84 inch of water at the design flow rate.

Outlet-header losses.- As shown in figure 4(b), the gas outlet header consists of a combination wye and a contraction. This shape is too irregular to lend itself to elementary analysis. It will be noted, however, that a 90° turn occurs in

the branch to turbosupercharger 1. If, under the condition of operation when the flow is equally divided between the two turbosuperchargers, the total pressures are the same in each branch upstream of the elbow, then any difference downstream should be the result of the turning loss in the elbow. This is the only loss that was calculated for the gas outlet header.

A flow rate of 6,000 pounds per hour through the elbow corresponds to a total flow rate of 12,000 pounds per hour through the heat exchanger, and for an R/D of 0.75 in the elbow the loss is 35 percent of the velocity head - or 0.32 inch of water.

Over-all losses.- The over-all pressure loss on the gas side has been calculated only for the condition where the flow is equally divided between the ducts leading to the two turbosuperchargers. This value for the branch of the system leading to turbosupercharger 2 is the sum of the inlet-header and heat-exchanger losses, and for the branch leading to turbosupercharger 1 is the sum of the inlet-header, heat-exchanger, and outlet-elbow losses. The two values are 1.71 and 2.05 inches of water, respectively, at the design flow rate.

The calculated values of pressure losses in the gas-side ducting are plotted in figure 6.

DESCRIPTION OF TEST APPARATUS

The heat exchanger and headers were assembled and tested for isothermal pressure losses in the arrangements shown in figures 7 to 10. As seen in these figures, a number of different arrangements were tested. It was possible in this manner to determine not only the pressure losses in the various parts, but also the magnitude of the interference effects.

The butterfly valve upstream of the venturi meter in the duct to turbosupercharger 1 (fig. 10) was shown by the tests to have a negligible effect on the accuracy of the venturi under the conditions of the tests.

The pressure rakes and traversing shielded total- and static-pressure tubes shown in figure 11 were used in making the pressure measurements. The manometers on which these pressures were indicated are shown in figure 8. In all cases, the multiple-tube manometers were used with the pressure rakes and micromanometers with the traversing tubes shown in figure

11(b). These micromanometers are mechanically driven and are set up to indicate a single differential pressure to a least count of 1 millimeter of fluid (in this case, water). The use of multiple-tube manometers, together with the pressure rakes, made it possible to obtain instantaneous flow patterns on the entire diameter of the ducts. The shielded total pressure tubes were used in regions where the direction of flow was uncertain. These tubes will indicate the correct total pressure when yawed through angles up to 60° from the direction of flow.

TEST PROCEDURE

The measured values of pressure losses in the heat-exchanger installation were obtained from the following arrangements. The general purpose of the procedure was to obtain as many cross checks on the data as were practical.

(a) The arrangements shown in figure 8 were used in obtaining the data on over-all pressure losses. It was in this form that the installation was originally designed.

(b) The losses in the heat exchanger were determined with the arrangements shown in figure 7, the straight headers allowing measurement of the losses as unaffected by the headers.

(c) Figures 9(a) and 10(a) show the arrangements in which the header losses were measured with the heat exchanger in place. These arrangements are identical with those in figures 8(a) and 8(b) except that traversing sections have been inserted between the headers and the heat exchanger.

(d) Figures 9(b) and 10(b) show arrangements used in measuring header losses with the heat exchanger removed.

The above-outlined procedure has made it possible to use the data from two independent tests in plotting the values of pressure loss in each part. In the case of the exhaust-gas side of the installation, two conditions of operation were investigated. Referring to figure 10, under one condition the branch to turbosupercharger 1 was blocked off and all air was directed to turbosupercharger 2, and, under the second

condition, the air flow was equally divided between the two turbosuperchargers. The over-all pressure losses were measured under both conditions. The radial locations at which pressure measurements were obtained in the round ducts are given by the rake details in figure 11(a). For the rectangular ducts in which the traverses were made, the duct cross section was assumed to be divided into 25 equal-area squares and pressure measurements were taken at the center of each square.

TEST RESULTS

The experimentally determined values of pressure losses in the air side of the installation are presented in figure 5 and those in the gas side in figures 6(a) and 6(b). The difference between the pressure losses determined with arrangements (c) and (d) in the test procedure was negligible and test points from both arrangements are presented in the curves in these figures.

Each test point represents the difference between average total pressures at stations upstream and downstream of the duct component. The average pressure at a station was obtained from 20 total-pressure readings in the case of the round ducts (for locations, see fig. 11(a)) or from 25 total-pressure readings at the centers of as many equal-area rectangles in the rectangular ducting.

In order to show the effect of the headers on the pressure loss in the air and gas passages of the heat exchanger, additional data were taken using the arrangements shown in figures 9(a) and 10(a). These data are compared in figure 12 with those taken using the arrangement in figures 7(a) and 7(b). Likewise, to show the effect of the traversing sections (which were installed between the headers and heat exchanger to measure the header losses) on the over-all pressure loss, data taken from the arrangements shown in figures 8(a), 8(b), 9(a), and 10(a) are plotted in figure 13.

ACCURACY OF MEASUREMENTS

From a consideration of the readability of the manometers, accuracy of venturi constants, and flow conditions in the installation, the flow rates are thought to be accurate to within ±2 percent, and the values of over-all pressure losses within about ±5 percent of the measured values.

Because of large-scale turbulence coupled with complete reversals of flow in the regions between the headers and heat exchanger, the accuracy of pressure-loss measurements in the headers is believed to be of the order of ±10 percent of the measured values.

DISCUSSION

A comparison of the analytical and experimental results in figures 5, 6(a), and 6(b) indicates that the predicted values of the losses were, in general, slightly low. The agreement being generally closer than 20 percent indicates that the analysis is applicable and is evidence of the fact that the interference effects did not cause a large increase in the losses in this installation over the predicted values in which the interference was assumed negligible.

The experimental and analytical results from figures 5, 6(a), and 6(b) are presented in bar-graph form in figures 14 and 15 for comparison of the losses in the component parts. The agreement between the experimental and analytical values for the over-all pressure loss is better than those for the component parts because of the more accurate measurement of over-all pressures.

When the values of measured pressure losses in the component parts of the gas side of the installation are added together, the summation exceeds the measured value of over-all pressure loss by the amount shown in figure 15 as experimental error in measuring component losses. This difference is attributed to the difficulty of making accurate measurements in the regions between the gas headers and heat exchanger. A similar difference between the summation of individual and over-all losses is not evident in the air ducting (fig. 14) where the large separation effects are not present and resulting measurements are more accurate.

The difficulty in obtaining accurate measurements in the region between the headers and heat exchanger is further emphasized by the comparison of gas-side data for the two conditions shown in figure 12 where the installation headers appear to reduce the pressure loss in the gas side of the heat exchanger. Such an effect is improbable and this difference is attributed to inaccuracies in measurement.

A header of the type shown in figure 4(a) is undesirable, not only because of the excessive pressure loss (in fig.

15 the pressure loss in the inlet header is shown to be larger than that in the heat exchanger), but also because of the uneven distribution of flow into the heat exchanger. Flow distribution in the various parts of the installation is shown in figures 16 to 19. It is to be noted that, while the heat exchanger has an appreciable damping effect on the distribution, figure 18 shows the uneven distribution caused by separation in the inlet header to persist into the region downstream of the heat exchanger.

One of the values in this type of analysis is in the location of losses which cannot be isolated experimentally. The test results can only indicate the magnitude of the loss through the duct component, and offer no clue to the cause of the loss or the distribution of the loss among several causes. Thus, without an analytical approach, the large contribution (fig. 14) of the expansion losses in the exchanger to the over-all loss across the exchanger might not be suspected. The corrugation expansion losses in the air side, which are not present in the gas side, plus the increased friction resulting from narrower gaps on the air side combine to cause the air-side loss to be over five times as large as the gas-side loss.

The agreement between the analytical and experimental values of over-all pressure loss on both sides of the installation shows that it is possible to predict analytically the magnitude of the losses in a ducting system of this type. However, it is obvious that there is a need for investigations of interference effects, particularly in expanding elbows of the type appearing in this installation.

In the ducting arrangements used in the present investigation, the air was drawn into the installation through short bell-entry ducts. Air delivered to the test region through such entries has a low-turbulence level and there is little opportunity for the growth of boundary layers at the walls. Several investigators have shown that when the flow is more disturbed and when thick boundary layers are present in the supply duct, the losses, particularly in diffusers, are increased because of the more favorable conditions for the occurrence of separation. It is probable that the inlet-header losses in this installation would increase under such conditions; however, it is not possible at the present time to predict their magnitudes. This points to the need for systematic investigations of the effects of initial turbulence and flow conditions.

CONCLUSIONS

1. Isothermal pressure losses can be predicted analytically in the type of ducting system which may be broken down into the elementary forms of elbows, diffusers, and straight ducts for which pressure-loss data are published.

2. For duct installations similar to that tested, in which the interference effects may be reasonably assumed to be small or negligible, the summation of the calculated losses for the individual components will be in close agreement with experimentally determined values of the over-all pressure loss.

3. The large-scale separation occurring in poorly designed expanding elbows results in uneven flow distribution and increased pressure loss. The results of this investigation indicate that the losses in such elbows can be predicted analytically with a fair degree of accuracy, but experimental values of the losses are subject to the uncertainties of pressure measurements in regions of uneven flow distribution.

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Moffett Field, Calif.

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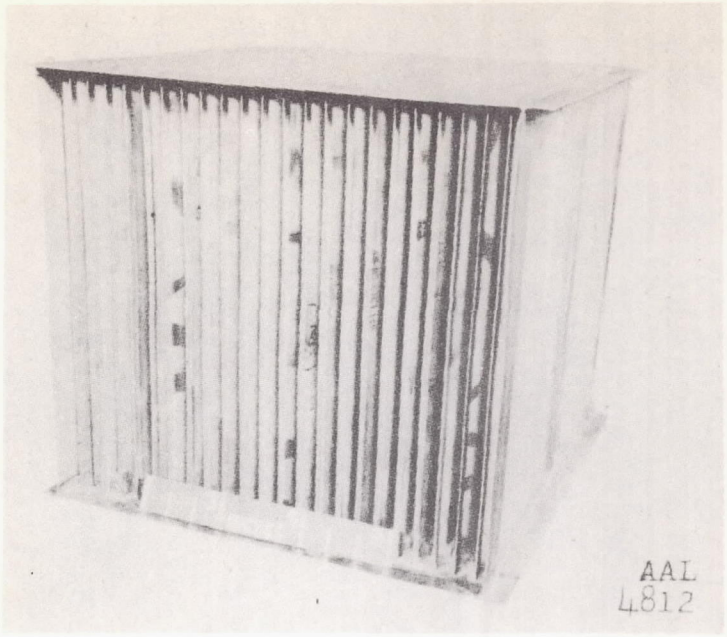
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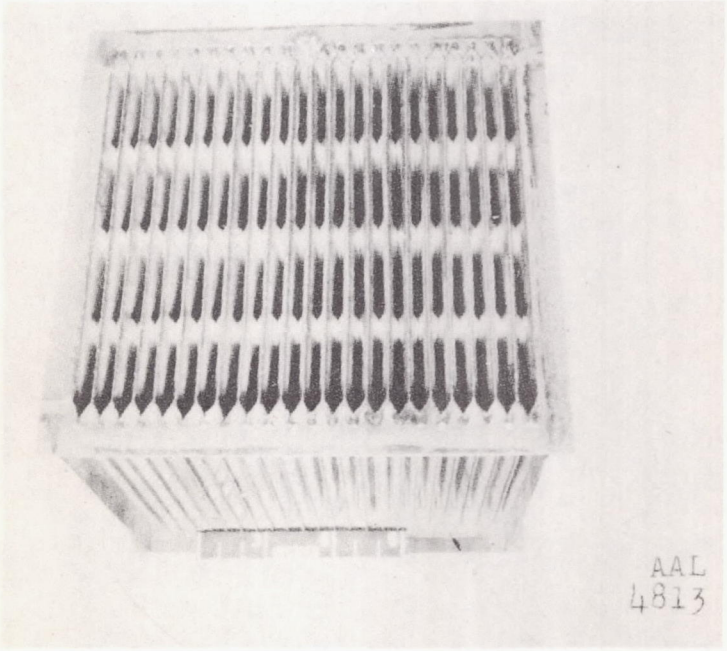
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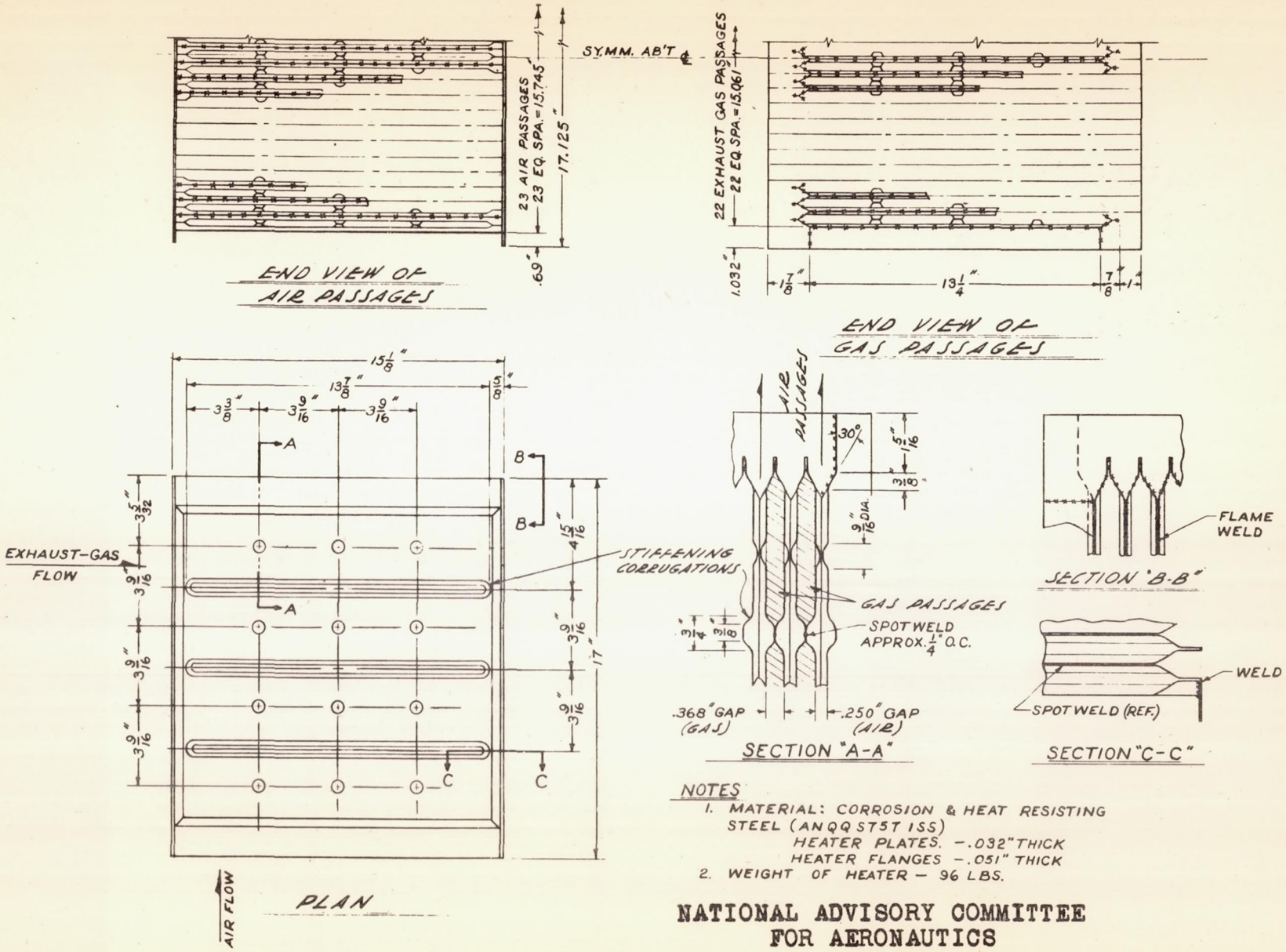


(a) Air passages



(b) Gas passages

Figure 1.- The exhaust-gas-to-air heat exchanger.

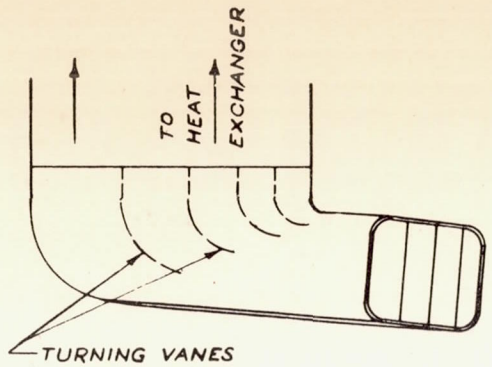


NOTES

1. MATERIAL: CORROSION & HEAT RESISTING STEEL (AN QQ S 75T 1SS)
 HEATER PLATES. -.032" THICK
 HEATER FLANGES -.051" THICK
2. WEIGHT OF HEATER — 96 LBS.

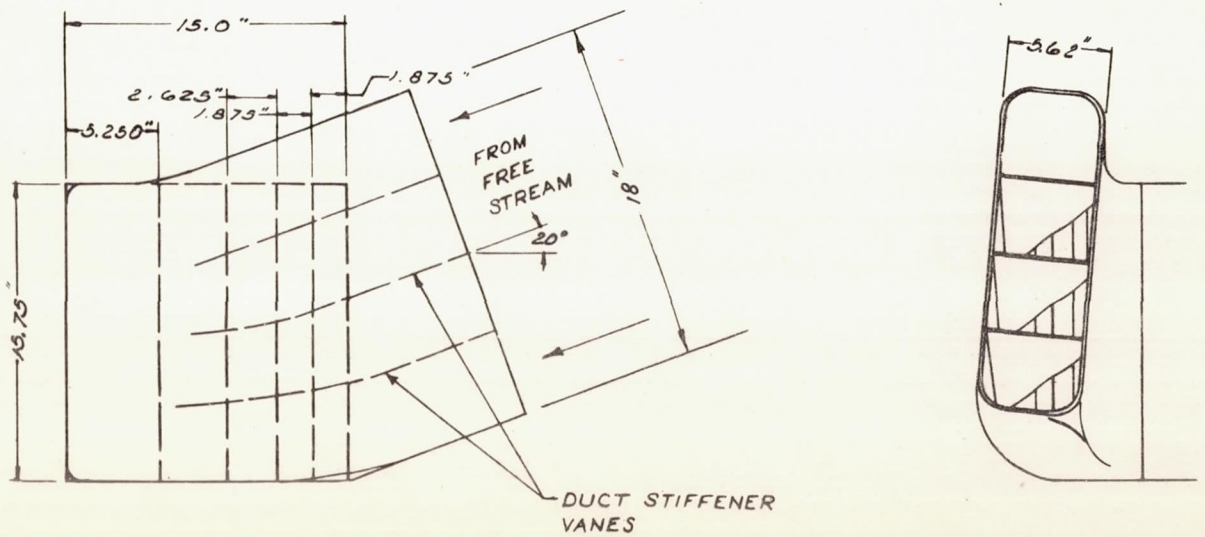
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FIGURE 2.-DETAILS OF THE HEAT EXCHANGER



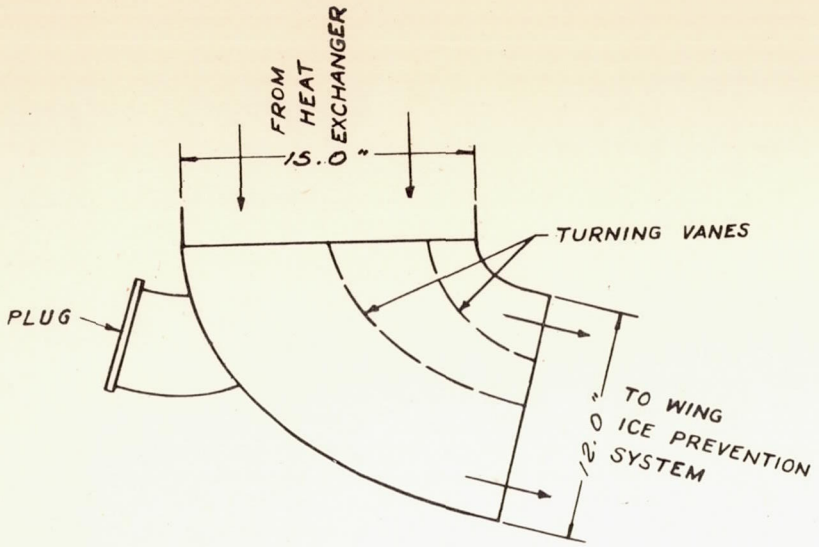
NOTE: ARROWS INDICATE DIRECTION OF FLOW

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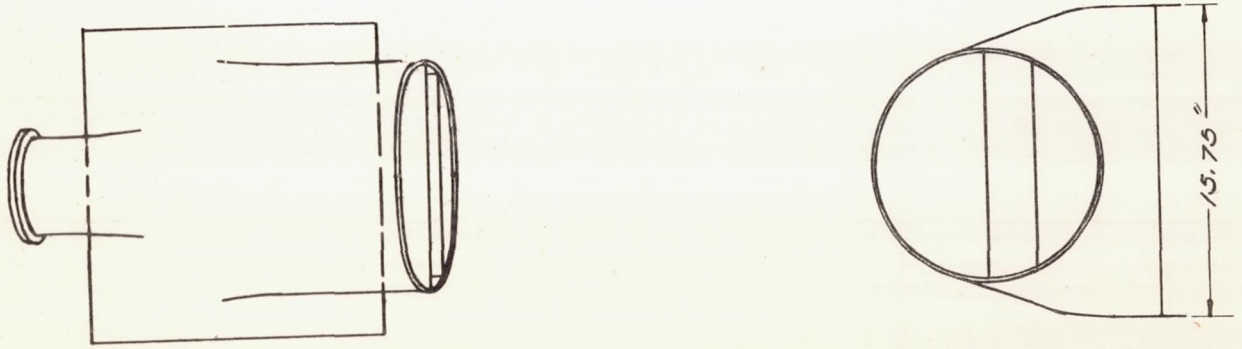
(a) INLET HEADER

FIGURE 3. - HEADERS WHICH DUCT AIR INTO AND OUT OF THE HEAT EXCHANGER



NOTE: ARROWS INDICATE DIRECTION OF FLOW

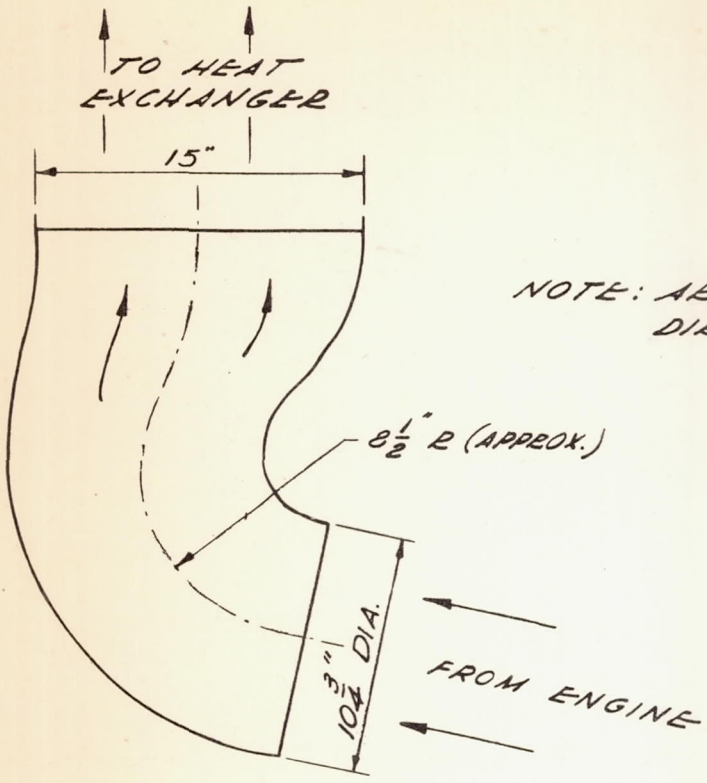
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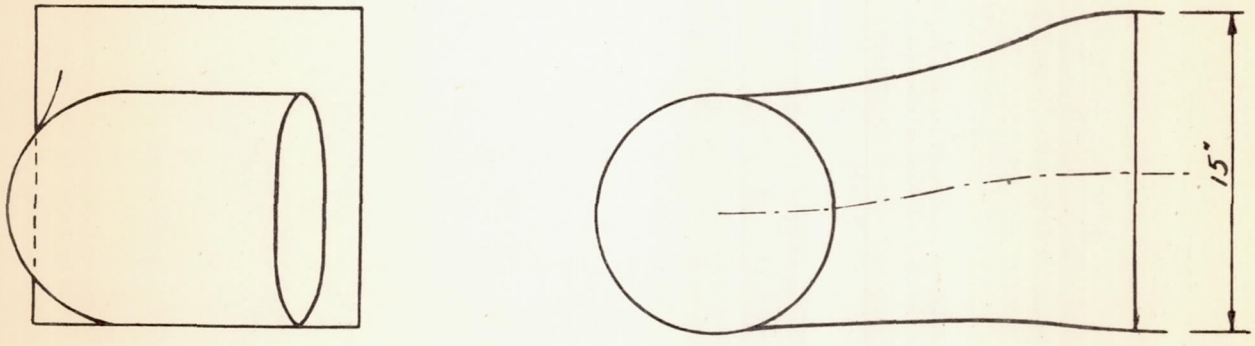
(b) OUTLET HEADER

FIGURE 3.- CONCLUDED

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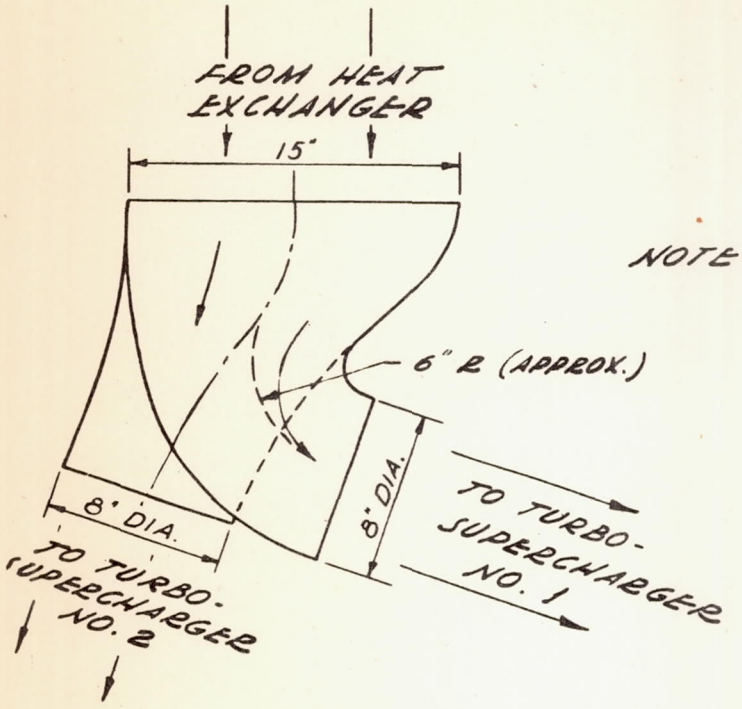
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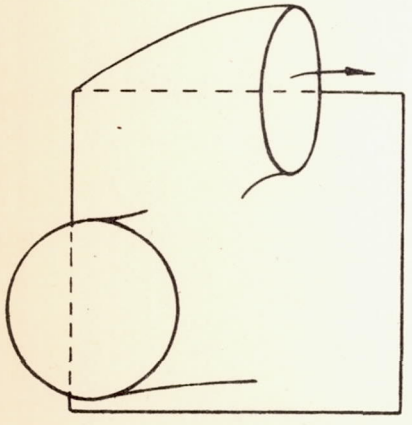
(a) INLET HEADER

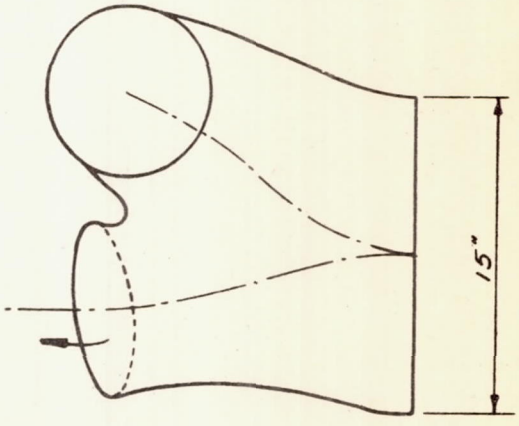
FIGURE 4. - HEADERS WHICH DUCT EXHAUST GAS INTO AND OUT OF THE HEAT EXCHANGER

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NOTE: ARROWS INDICATE DIRECTION OF FLOW





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(6) OUTLET HEADER

FIGURE 4. - CONCLUDED.

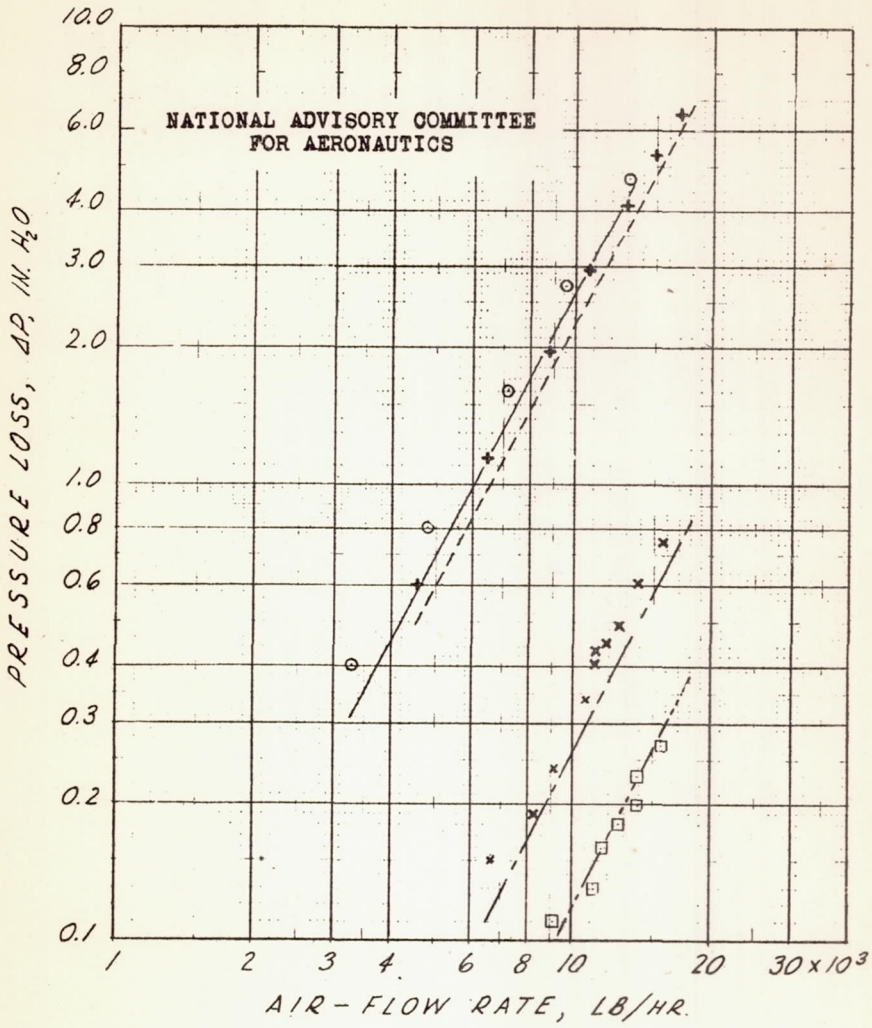
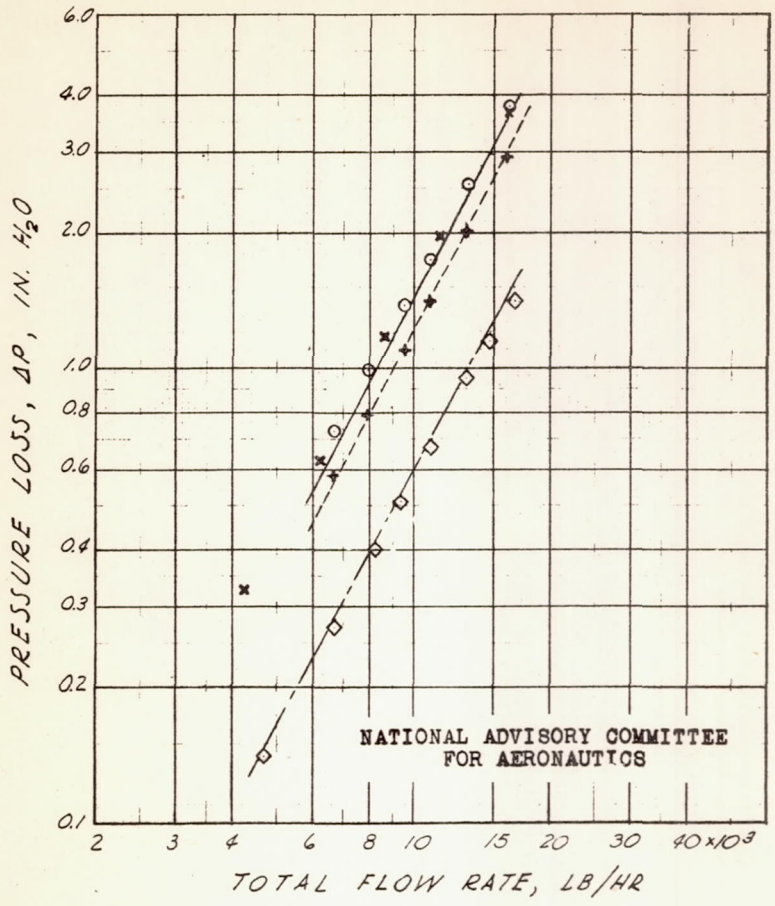


FIGURE 5.- CALCULATED AND MEASURED VALUES OF PRESSURE LOSSES IN THE AIR SIDE OF THE HEAT-EXCHANGER INSTALLATION AS A FUNCTION OF AIR-FLOW RATE.

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——— CALCULATED OVERALL PRESSURE LOSS TO TURBOSUPERCHARGER Nº 1 } FLOW EQUALLY DIVIDED

----- CALCULATED OVERALL PRESSURE LOSS TO TURBOSUPERCHARGER Nº 2 } FLOW EQUALLY DIVIDED

——— CALCULATED HEAT EXCHANGER PRESSURE LOSS.

o MEASURED OVERALL PRESSURE LOSS TO TURBOSUPERCHARGER Nº 1 } FLOW EQUALLY DIVIDED

\+ MEASURED OVERALL PRESSURE LOSS TO TURBOSUPERCHARGER Nº 2 } FLOW EQUALLY DIVIDED

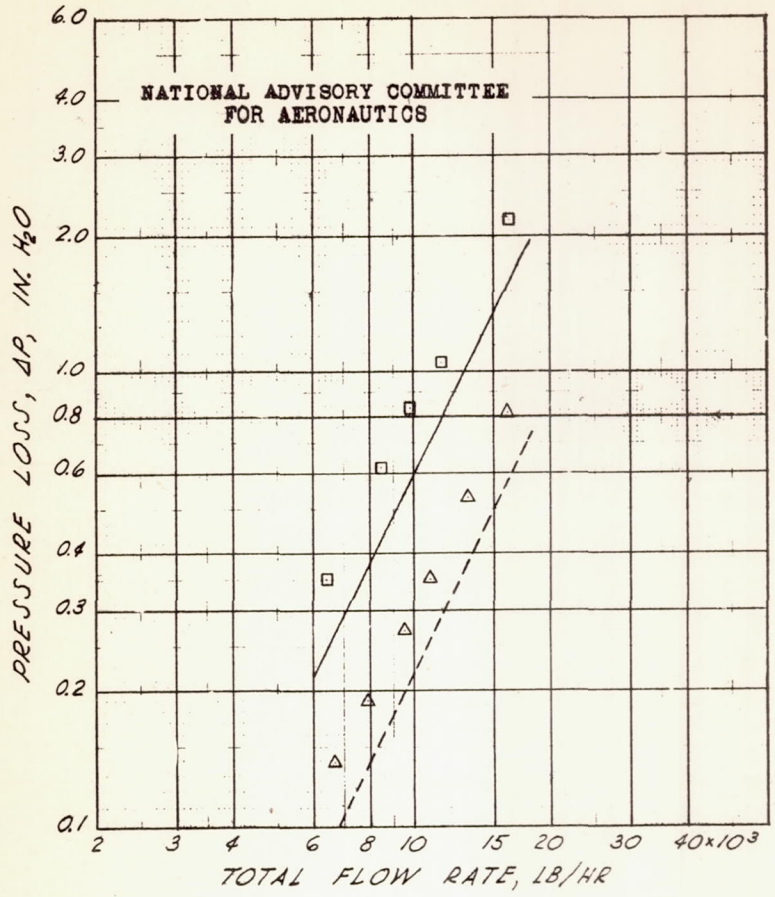
◇ MEASURED HEAT EXCHANGER PRESSURE LOSS.

× MEASURED OVERALL PRESSURE LOSS WITH TOTAL FLOW TO TURBOSUPERCHARGER Nº 2.

(a) OVERALL AND HEAT-EXCHANGER LOSSES

FIGURE 6 — CALCULATED AND MEASURED VALUES OF PRESSURE LOSSES IN THE GAS SIDE OF THE HEAT-EXCHANGER INSTALLATION.

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——— CALCULATED INLET HEADER PRESSURE LOSS.
- - - - - CALCULATED ELBOW LOSS IN BRANCH OF OUTLET HEADER TO TURBOSUPERCHARGER No. 1. FLOW RATE IN BRANCH EQUALS HALF OF TOTAL FLOW RATE.
□ MEASURED INLET HEADER PRESSURE LOSS.
△ MEASURED DIFFERENCE BETWEEN OVERALL PRESSURE LOSSES WITH FLOW EQUALLY DIVIDED BETWEEN TURBOSUPERCHARGERS.

(6) HEADER LOSSES

FIGURE 6. — CONCLUDED.

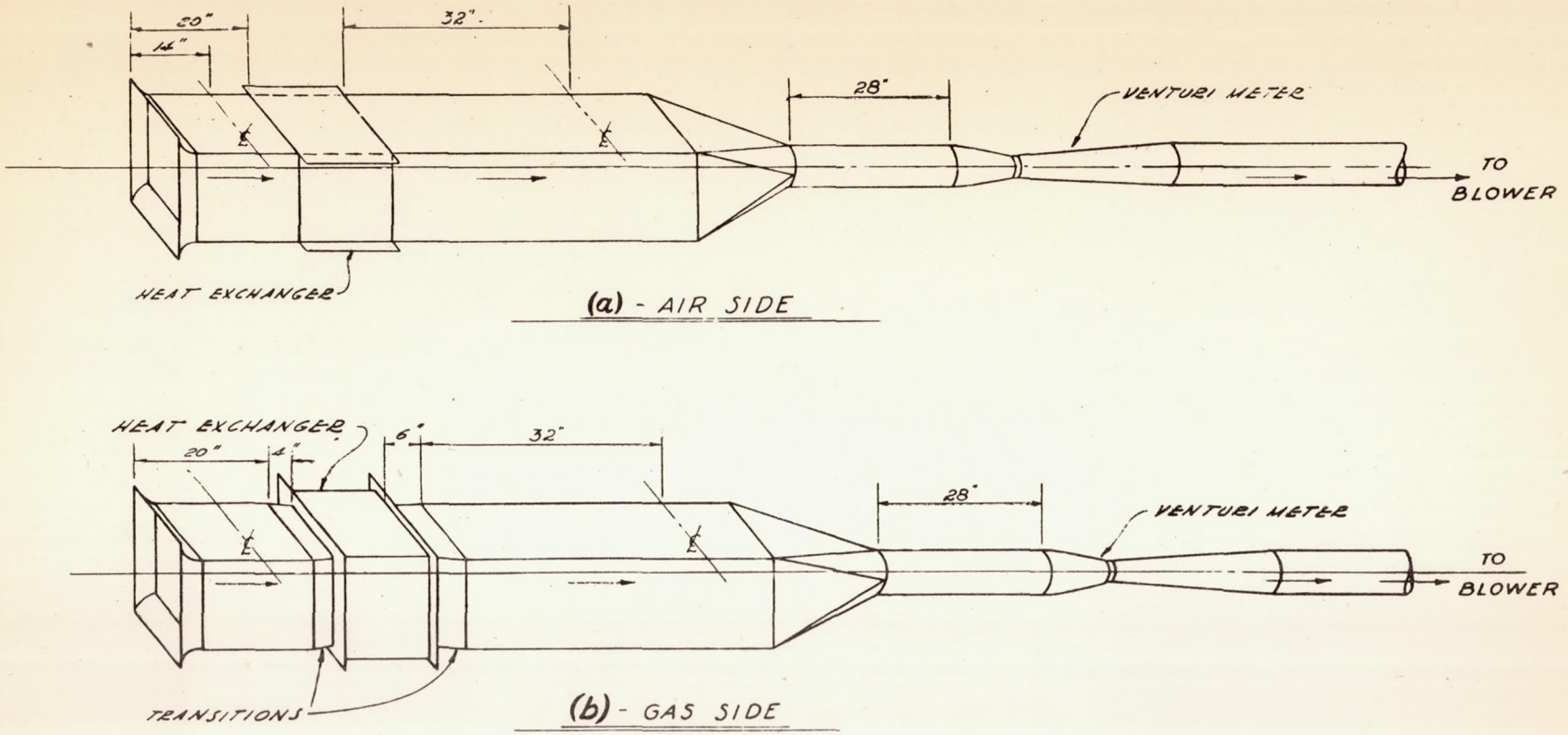
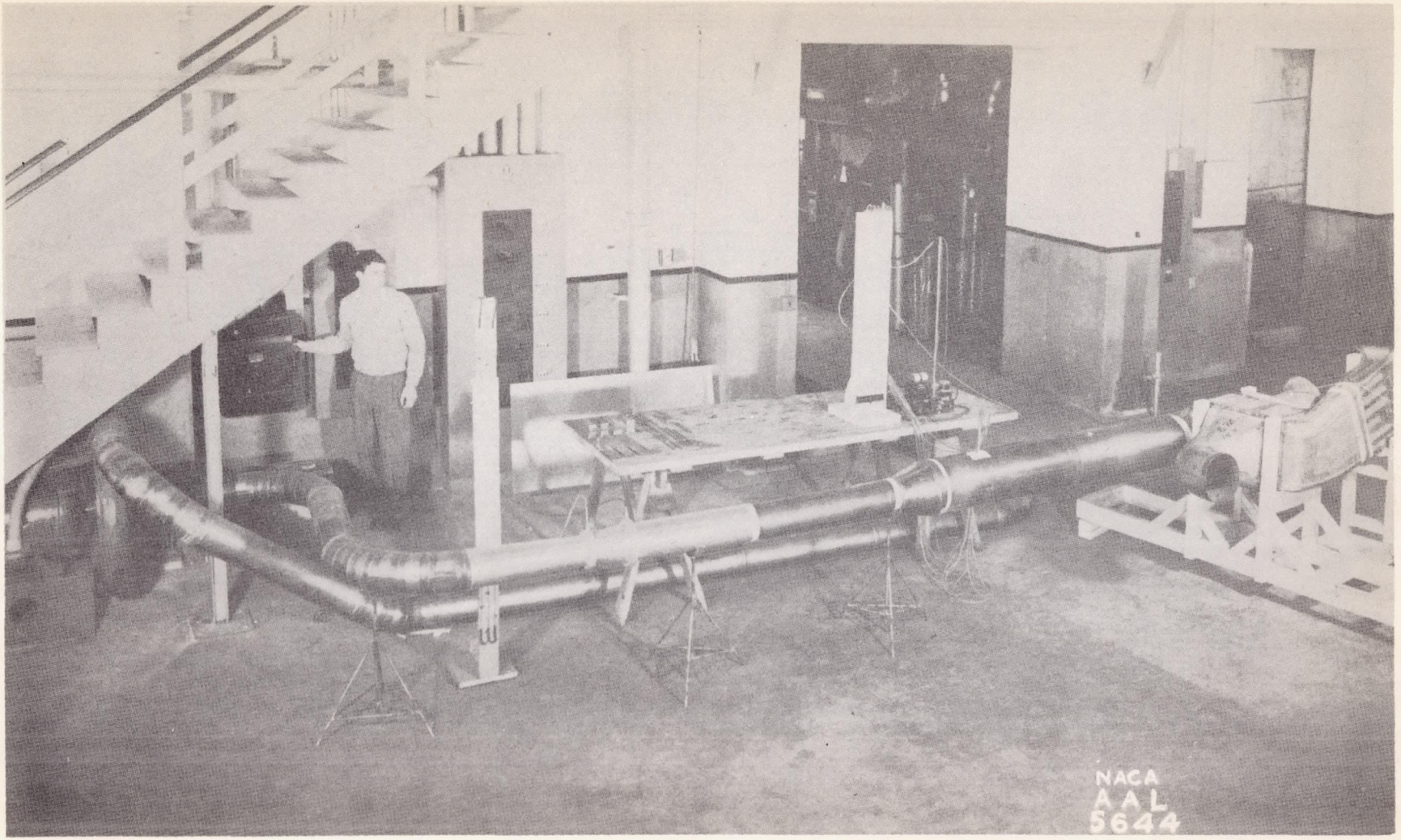
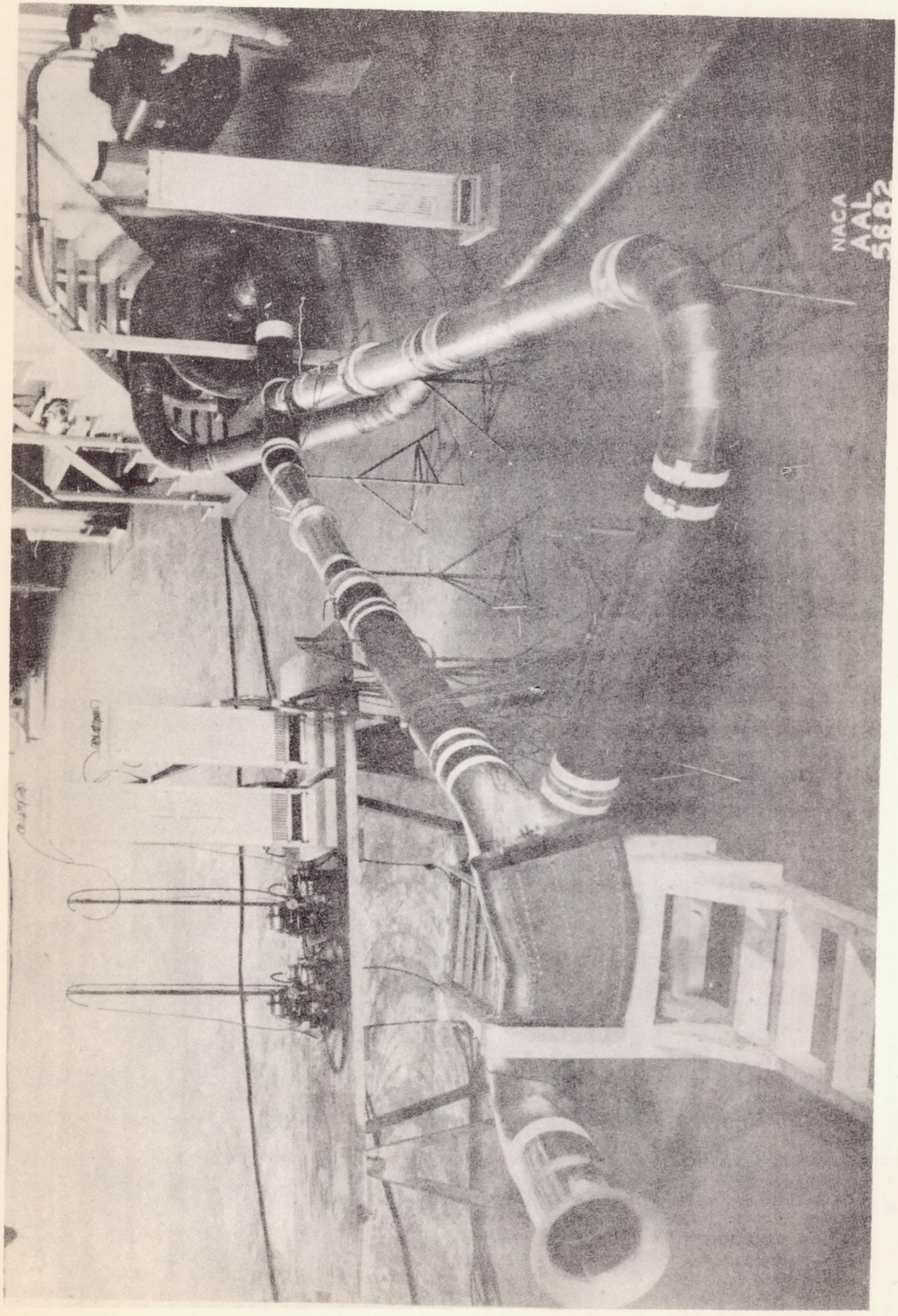


FIGURE 7. — LAYOUT OF DUCTING AND LOCATION OF PRESSURE TUBES FOR MEASUREMENT OF PRESSURE LOSS IN ISOLATED HEAT EXCHANGER



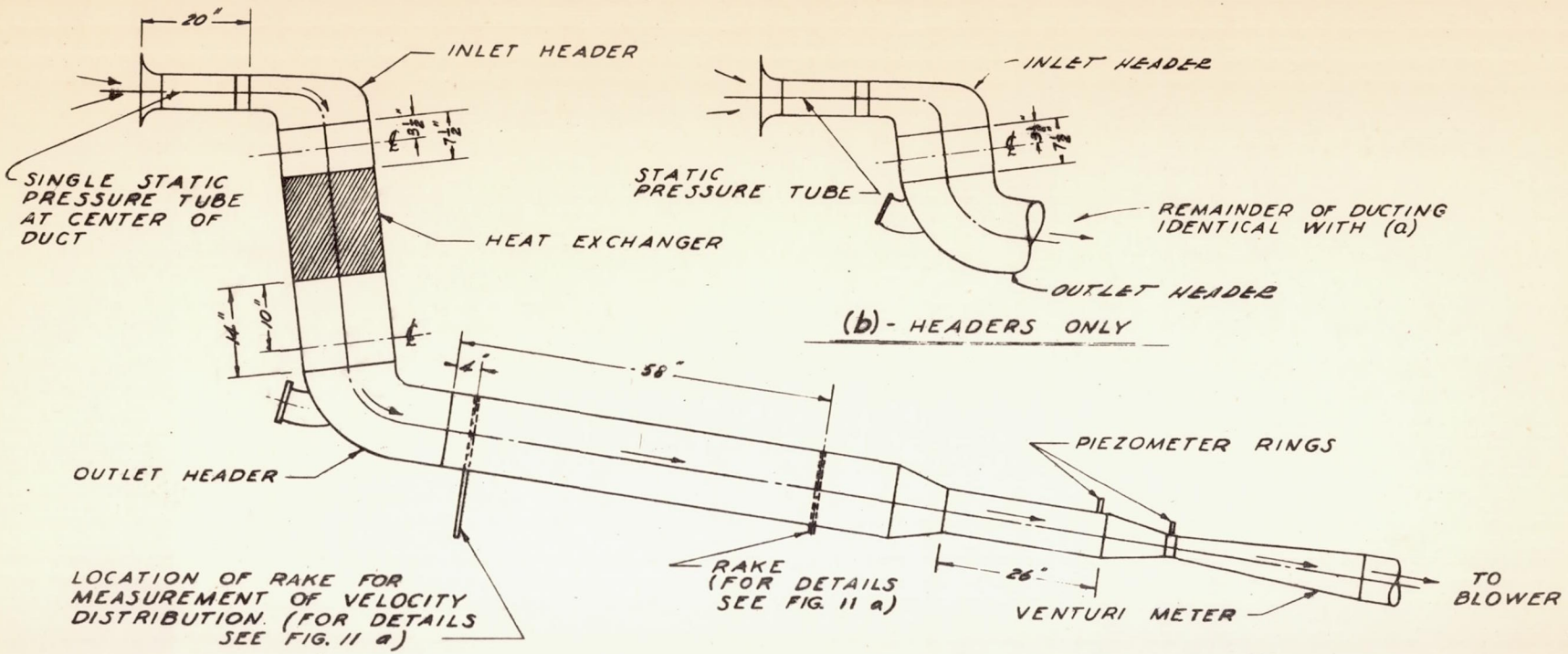
(a) Air side

Figure 8a,b.- Ducting and instrumentation for measuring the over-all pressure loss in the complete heat-exchanger installation.



(b) Gas side

Figure 8.- (Concluded).



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FIGURE 9. - LAYOUT OF THE DUCTING AND LOCATION OF THE INSTRUMENTATION FOR MEASURING THE PRESSURE LOSSES IN (a) THE PARTS OF THE AIR SIDE OF THE COMPLETE HEAT-EXCHANGER ASSEMBLY AND (b) THE AIR HEADERS.

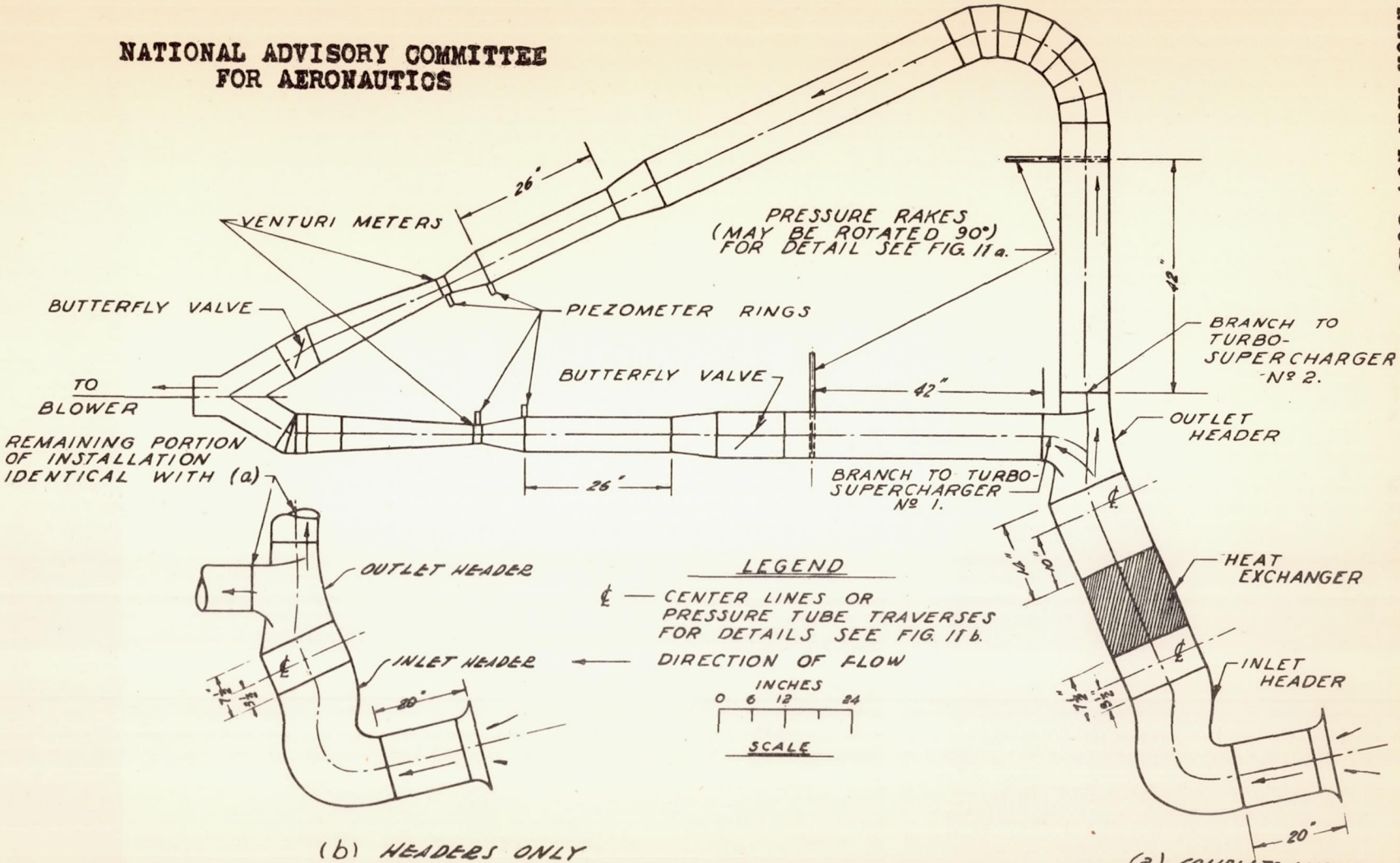
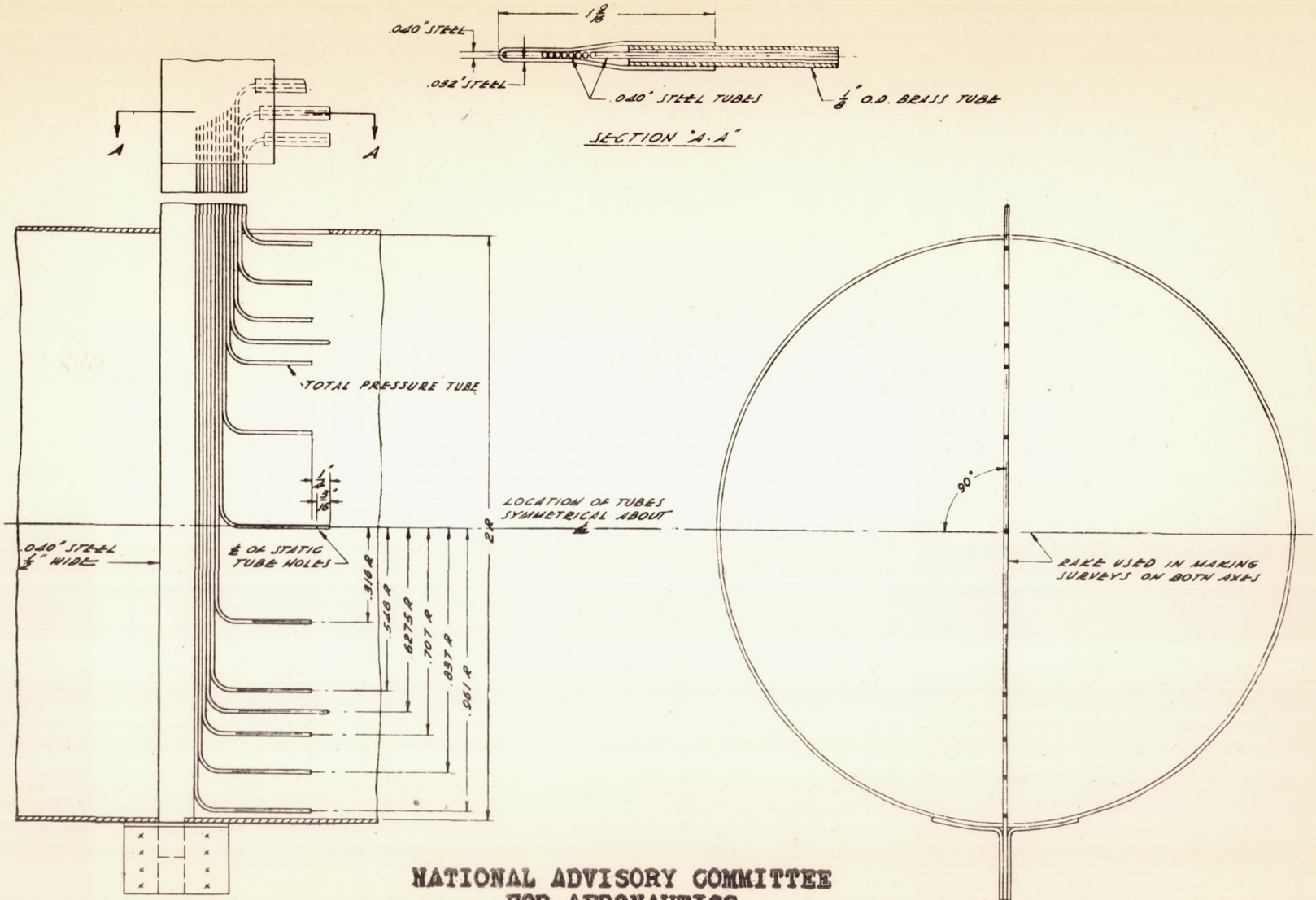
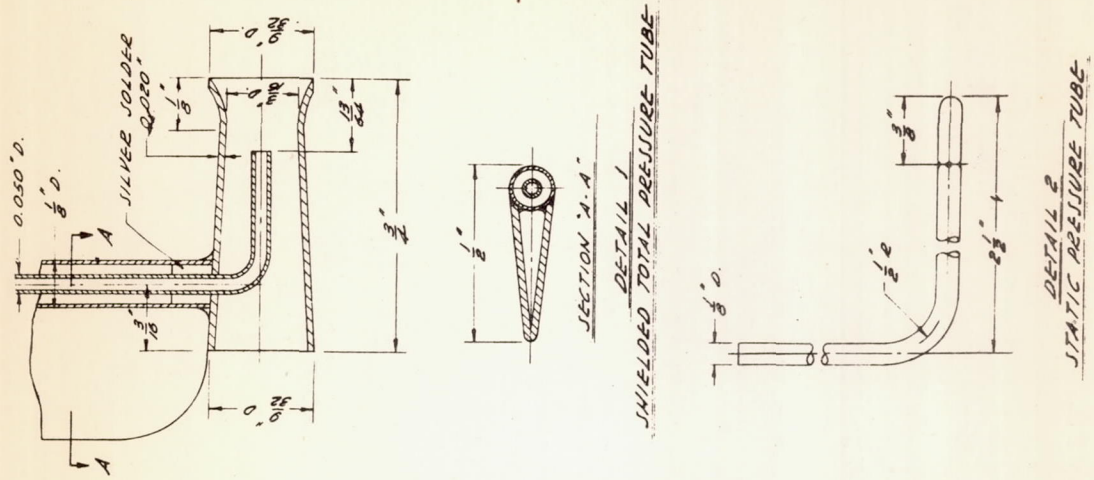


FIGURE 10. - LAYOUT OF THE DUCTING AND LOCATION OF THE INSTRUMENTATION FOR MEASURING THE PRESSURE LOSSES IN (a) THE PARTS OF THE GAS SIDE OF THE COMPLETE HEAT-EXCHANGER ASSEMBLY AND (b) THE GAS HEADERS.

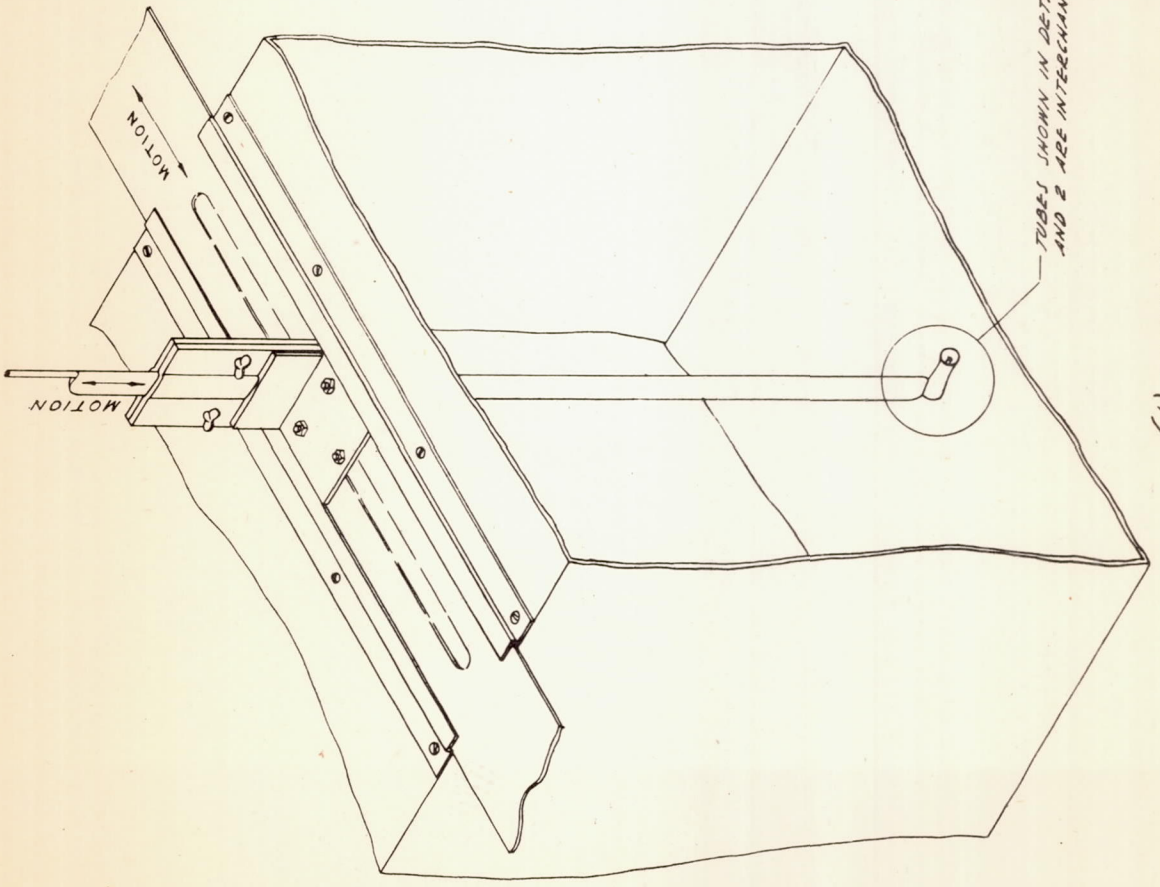


(3) PRESSURE RAKES

FIGURE 11. — INSTRUMENTATION DETAILS



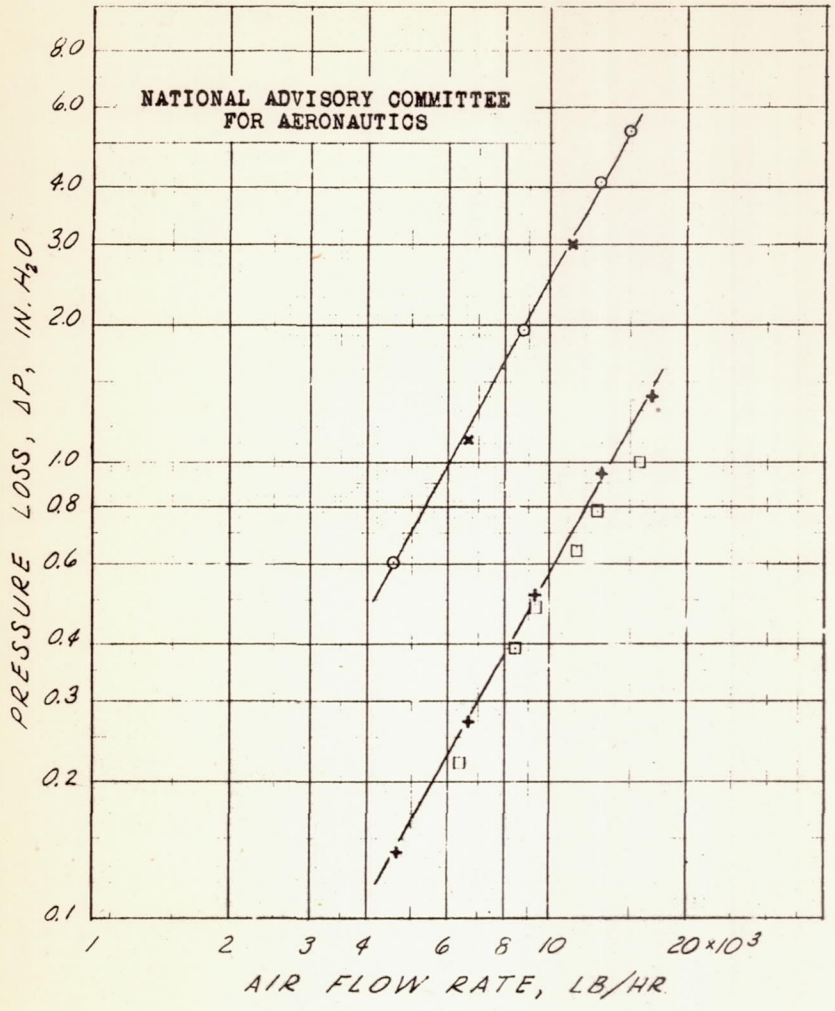
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(b) TRAVERSABLE SHIELDED TOTAL AND STATIC PRESSURE TUBES

FIGURE 11.- CONCLUDED

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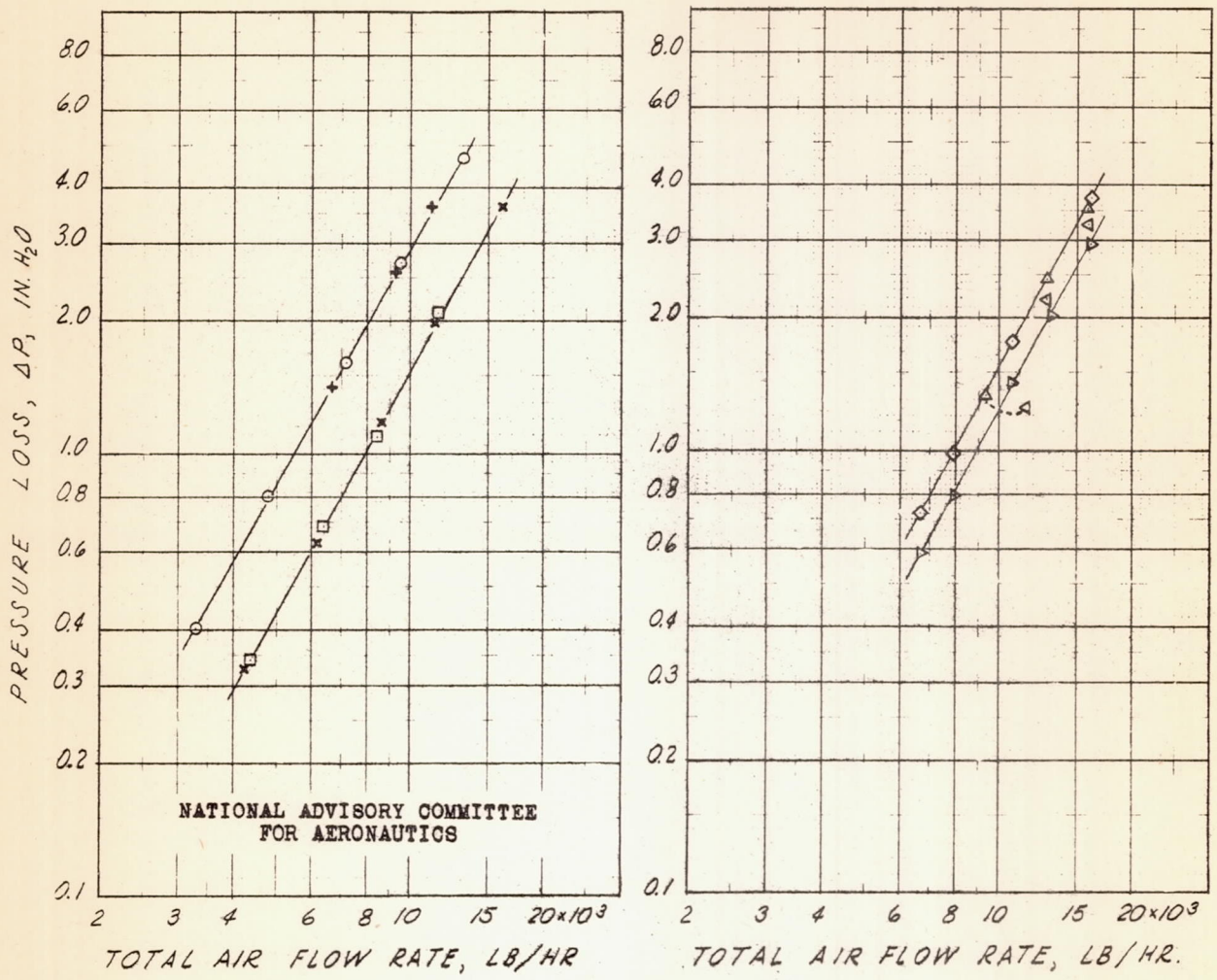


MEASURED PRESSURE LOSS IN HEAT EXCHANGER WITH STRAIGHT HEADERS (SEE FIG. 7)
- ○ AIR SIDE
- ✚ GAS SIDE

MEASURED PRESSURE LOSS IN HEAT EXCHANGER WITH INSTALLATION HEADERS (SEE FIGS 9a & 10a)
- ✖ AIR SIDE
- □ GAS SIDE

FIGURE 12.- COMPARISON OF MEASURED VALUES OF PRESSURE LOSS IN THE HEAT EXCHANGER WITH AND WITHOUT HEADERS AS A FUNCTION OF FLOW RATE.

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MEASURED OVERALL PRESSURE LOSS OF COMPLETE INSTALLATION ARRANGED AS SHOWN IN FIGURES 8a AND 8b

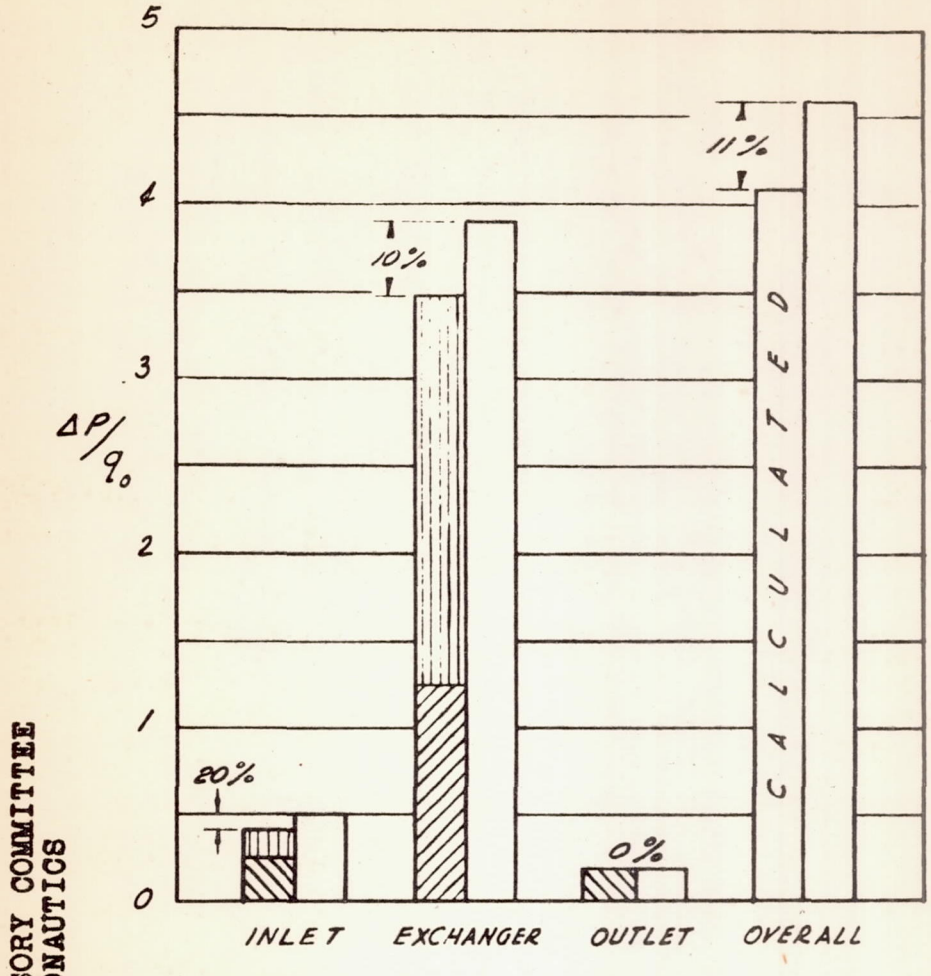
- ○ AIR SIDE
- ◇ GAS SIDE, TURBOSUPERCHARGER №1 } FLOW EQUALLY DIVIDED
- ▷ GAS SIDE, TURBOSUPERCHARGER №2 }
- × GAS SIDE, ALL FLOW TO TURBOSUPERCHARGER №2.

MEASURED OVERALL PRESSURE LOSS OF COMPLETE INSTALLATION EQUIPPED WITH TRAVERSING SECTIONS AS SHOWN IN FIGURES 9a AND 10a.

- \+ AIR SIDE
- △ GAS SIDE, TURBOSUPERCHARGER №1 } FLOW EQUALLY DIVIDED.
- ◁ GAS SIDE, TURBOSUPERCHARGER №2 }
- □ GAS SIDE, ALL FLOW TO TURBOSUPERCHARGER №2.

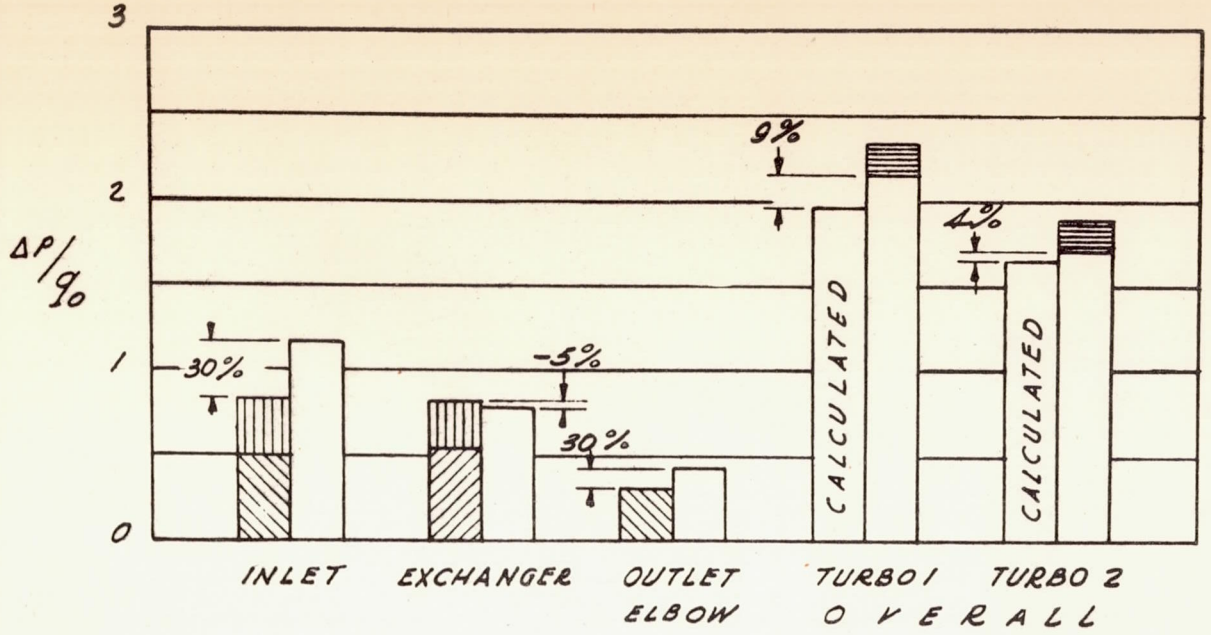
FIGURE 13.- COMPARISON OF MEASURED VALUES OF OVERALL PRESSURE LOSS THROUGH THE HEAT-EXCHANGER INSTALLATION WITH AND WITHOUT TRAVERSING SECTIONS BETWEEN HEADERS AND HEAT EXCHANGER.

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FIGURE 14.- COMPARISON OF CALCULATED AND MEASURED VALUES OF $\Delta P/q_0$ FOR THE AIR-SIDE PARTS OF THE HEAT-EXCHANGER INSTALLATION.



CALCULATED FRICTION LOSS.

CALCULATED ELBOW LOSS.

CALCULATED EXPANSION LOSS.

EXPERIMENTAL LOSS.

EXPERIMENTAL ERROR IN MEASURING COMPONENT LOSSES

$$\% = \frac{\text{EXPERIMENTAL} - \text{CALCULATED}}{\text{EXPERIMENTAL}}$$

INLET VELOCITY HEAD, q_0, = 1.03 IN. H_2O AT FLOW RATE OF 12,000 LB. PER HR.

FIGURE 15.- COMPARISON OF CALCULATED AND MEASURED VALUES OF $\Delta P/q_0$ FOR THE EXHAUST-GAS-SIDE PARTS OF THE HEAT-EXCHANGER INSTALLATION.

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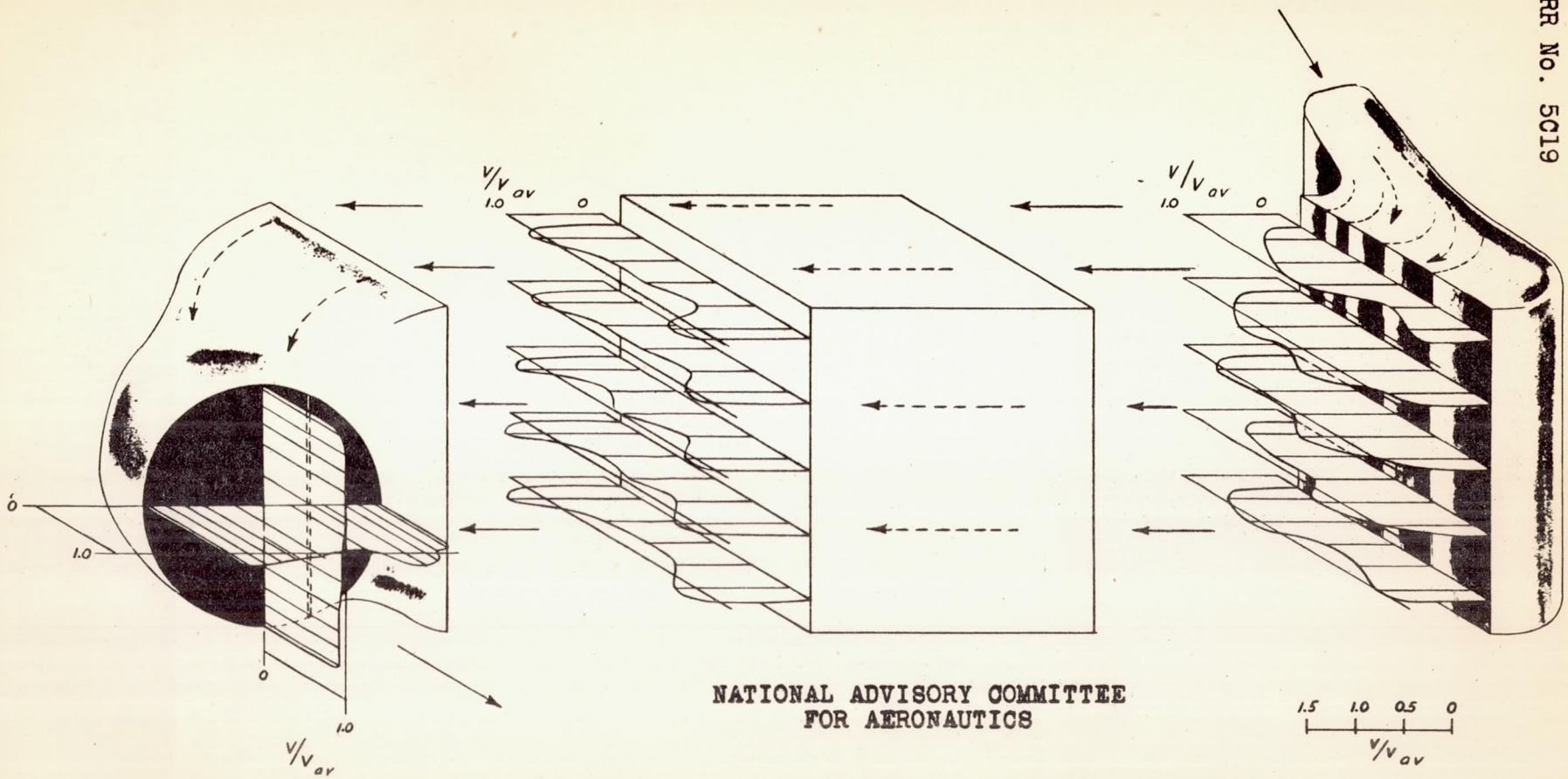


FIGURE 16.- VELOCITY DISTRIBUTIONS IN THE AIR SIDE OF THE COMPLETE HEAT-EXCHANGER INSTALLATION.

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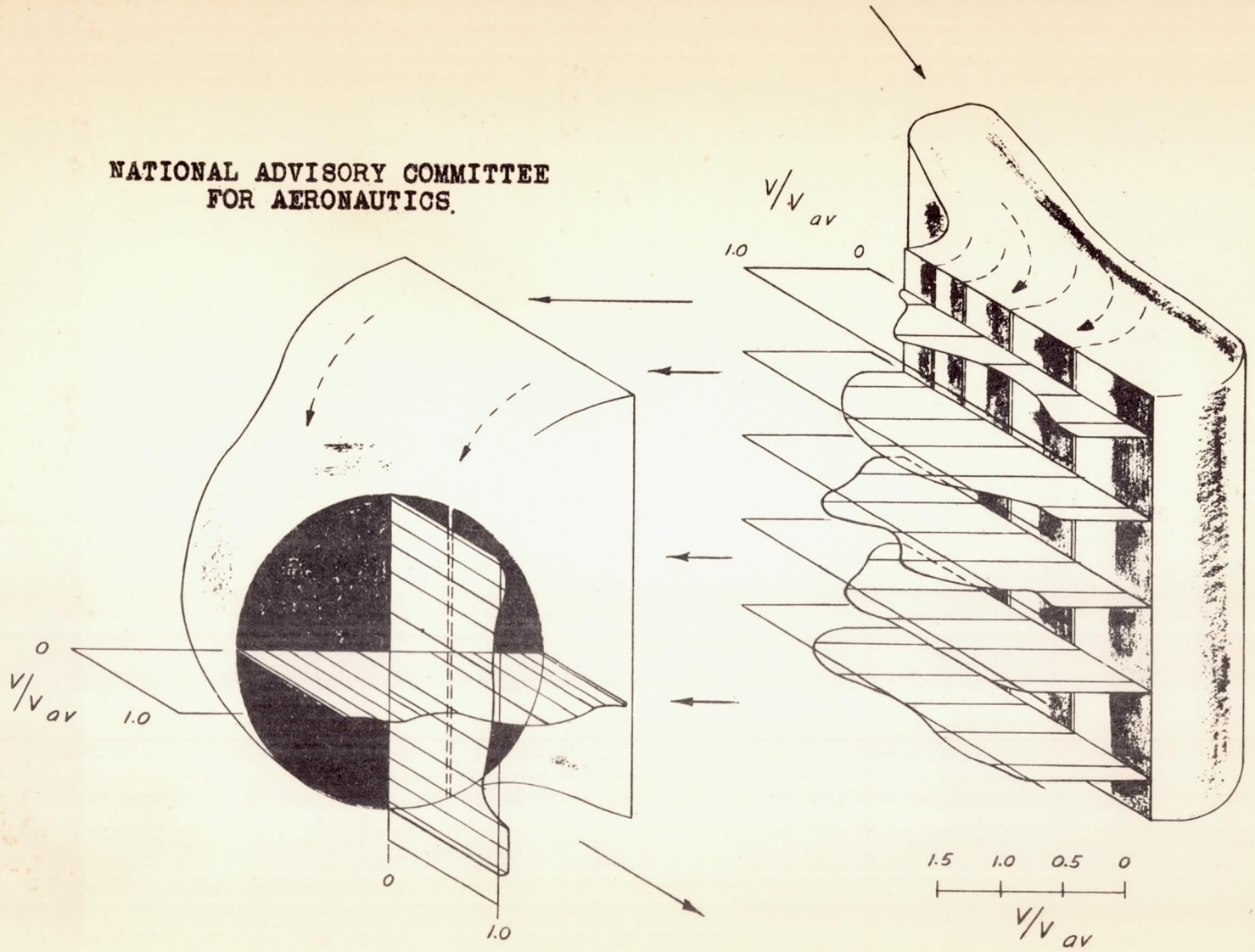


FIGURE 17. - VELOCITY DISTRIBUTIONS IN THE AIR HEADERS WITH THE HEAT EXCHANGER REMOVED

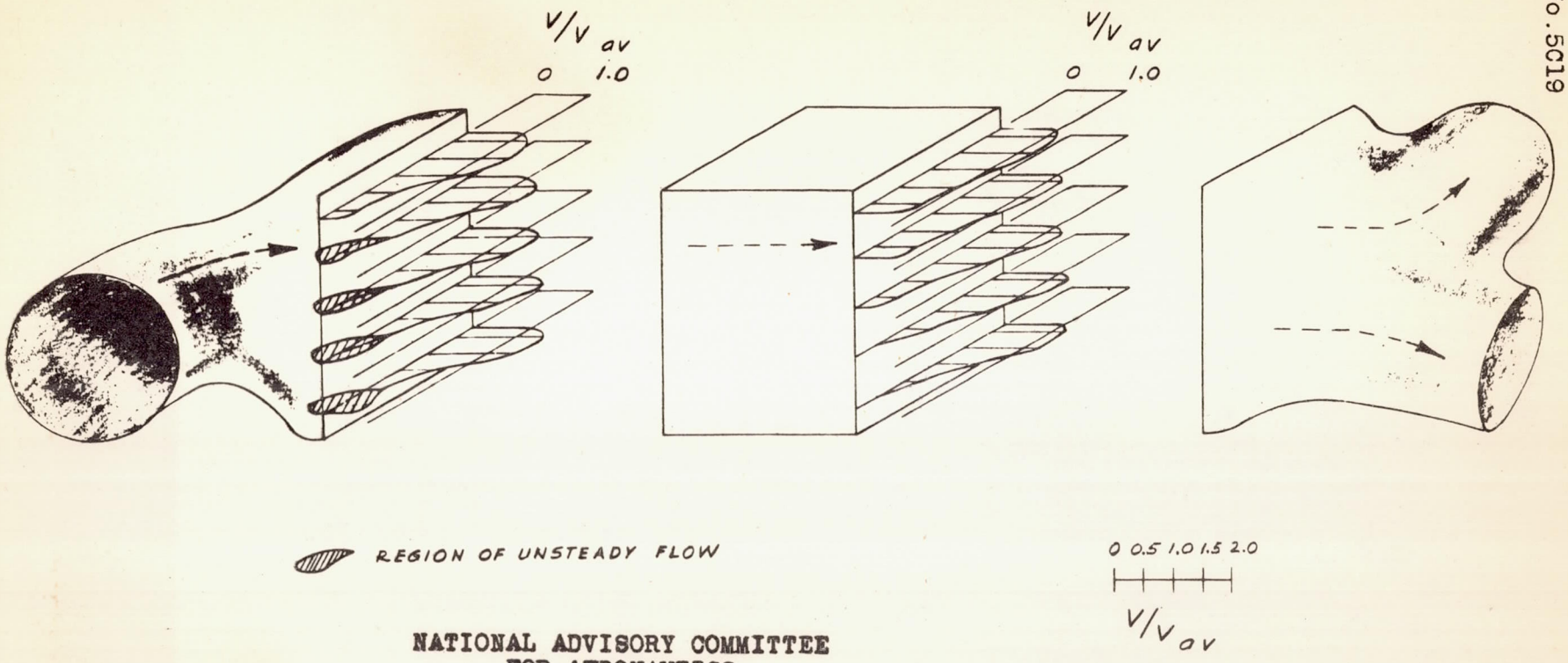


FIGURE 18.- VELOCITY DISTRIBUTIONS IN THE GAS SIDE OF THE COMPLETE HEAT-EXCHANGER INSTALLATION.

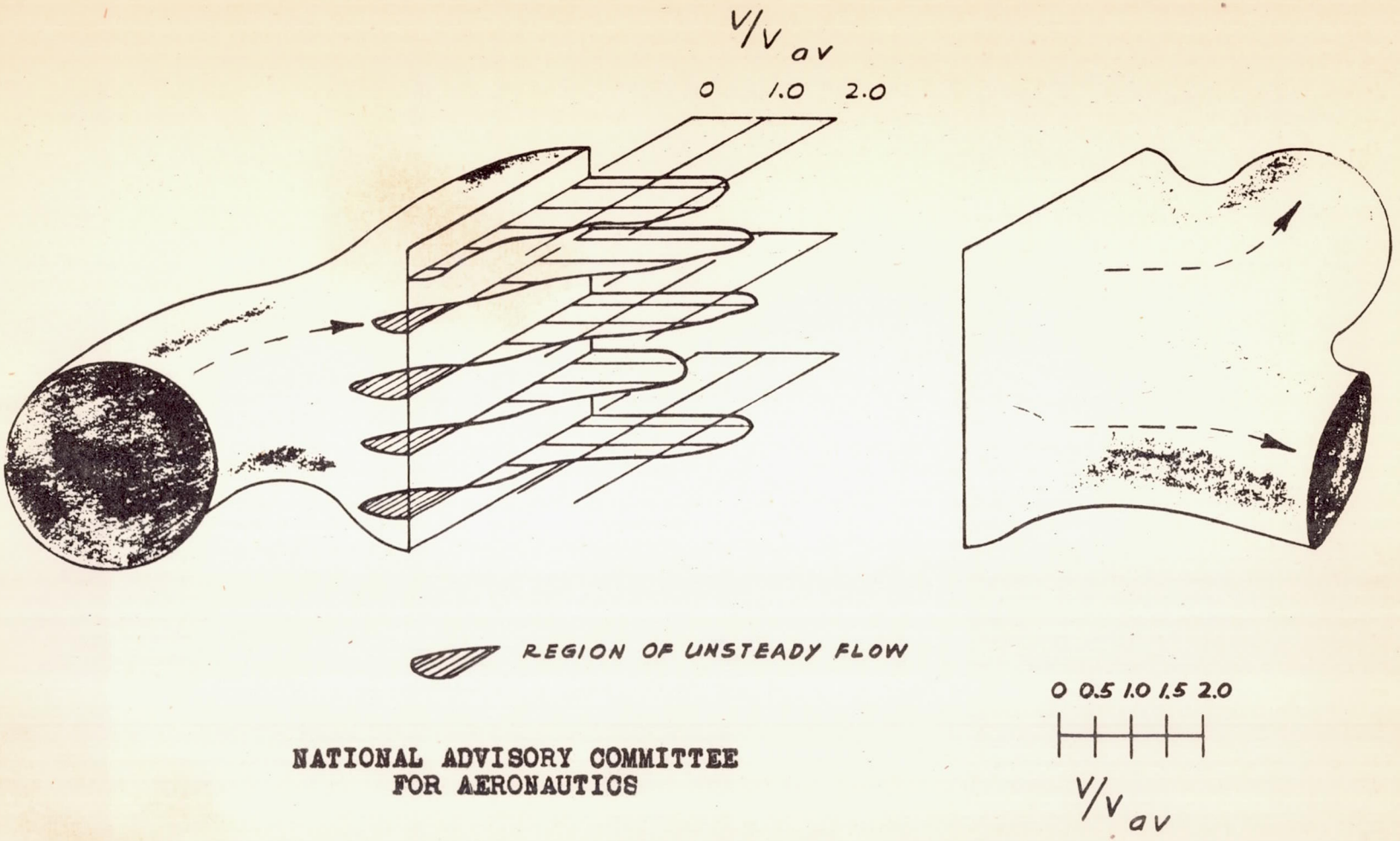


FIGURE 19.- VELOCITY DISTRIBUTIONS IN THE GAS INLET HEADER WITH THE HEAT EXCHANGER REMOVED.